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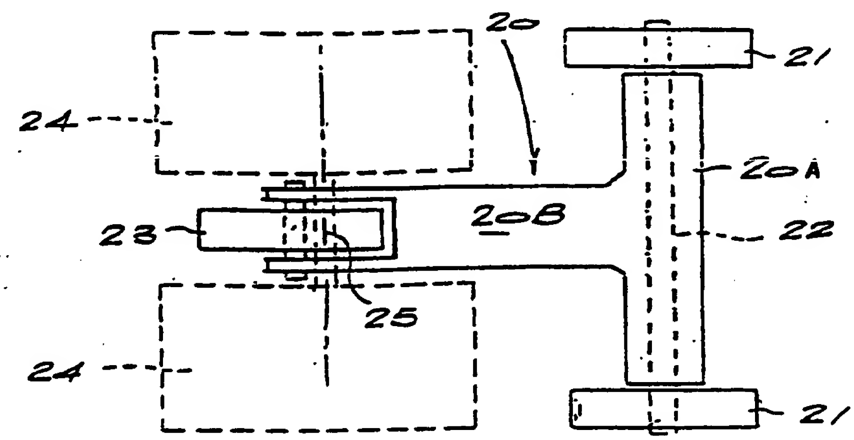
(54) Title: SOIL COMPACTOR WITH STABILISING WHEEL SYSTEM

(57) Abstract



The invention concerns a self-propelled impact compaction machine that includes a chassis (20) with a prime mover thereon. The chassis (20) carries ground engaging wheels (21) at least some of which are driven by the prime mover. Compaction of soil is achieved by one or more impact compactor masses (24) which have out-of-round shapes so as to apply periodic impact blows to the soil surface when rolled over that surface. The mass or masses (24) are connected resiliently to the chassis (20) and define a compaction track width measured from one lateral extremity of the mass or masses (24) to the opposite lateral extremity of the mass or masses. The ground engaging (21) wheels of the chassis define a maximum wheel track width which is not substantially different in magnitude to the compaction track width, with the desirable result that the chassis (20) enjoys substantial roll axis stability.

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"SOIL COMPACTOR WITH STABILISING WHEEL SYSTEM"

BACKGROUND TO THE INVENTION

THIS invention relates to soil compaction using a soil compactor of the general type first described in US patent 2,909,106.

The term "Impact Roller", as used initially in this US patent, refers to a soil compaction machine incorporating a compactor mass of non-round shape which, when towed over a soil surface, produces a series of periodic blows on the soil surface. The compactor mass of an impact roller has a series of spaced apart, salient points on its periphery. Each such salient point is followed by a re-entrant portion of the periphery and each re-entrant portion is followed in turn by a compacting face. As the impact roller is towed over the soil surface, for instance by means of a tractor, it rises up on each salient point and then falls forwardly and downwardly as it passes over that point, with the result that the following compacting face applies an impact blow to the soil surface.

The action of the compactor mass is to accumulate potential energy as the compactor mass rises up on each salient point, then to deliver this energy as an impact blow as the compactor mass falls and the compacting face strikes the soil surface.. The coupling between the tractor and the compactor mass is resilient in nature to allow for the necessary forward and downward falling motion undergone by the mass as it passes over each salient point.

Impact rollers incorporating a single compactor mass have been commercially built and successfully used by the civil engineering industry. Dual-mass impact rollers of the type described in European patent 0017511 have superseded the single mass type.

A three-sided dual-mass impact roller with its towing tractor is illustrated in side elevation in Figure 1, in which one of the masses has been removed to reveal the components situated between the compactor masses. Figure 2 shows a rear view of the impact roller, and Figure 3 a plan view thereof.

In Figures 1, 2 and 3 the carriage 1 is located between the two compactor masses 2, which are joined together by means of a common axle assembly 3. The axle assembly 3 consists of an outer tubular axle and an inner torsion bar, not shown, which keeps the masses rotating synchronously. The inner torsion bar is a resilient member in order to allow for a limited amount of twist between the masses.

The axle assembly 3 is connected by means of a hydraulically controlled linkage system to the carriage 1. The linkage system comprises a drag-link 4 fixed at one extremity to the axle 3, and pivotally connected at the other extremity to the upper extremity of a drop-link 5 such that only pitch axis rotation can occur.

The drop-link is pivotally connected at its lower extremity to the carriage 1 such that only pitch axis rotation can occur and at its upper extremity to a traction device 6 which in practice functions to provide both traction and damping. Two pairs of ground engaging wheels 7 are mounted on the carriage. Hydraulic lift cylinders 8 act between the carriage 1 and the axle

3 to lift the compactor masses off the ground when transporting the impact roller.

The carriage is connected to the towing tractor 10 by means of a universal coupling 9 which allows for relative movement between the tractor and the impact roller about all three axes. Thus the coupling 9 allows for roll, yaw and pitch of the impact roller, where "roll" refers to rotational movement of the impact roller about the fore-and-aft direction of travel.

Tractor towed impact rollers as illustrated in Figures 1, 2, and 3 are now in general use. The compactor masses may typically have three, four or five sided shapes, and together they generally weigh between three and ten tons.

Referring to Figure 4, the dimensions R and r could typically be 1100mm and 900mm respectively. The amount of potential energy stored as the mass rises on to a salient point is stated as M.g.h where h is equal to R-r (in metres) and M is the mass in kilograms. Thus for a pair of masses weighing 10 tons (10 000 kg), the energy available for each impact blow is 10 000 x (1100-900) x 9.81 = 19 600J, or 19.6kJ.

For an impact rolling machine of this specification, i.e. 19.6kJ, the width of the masses shown in Figure 3 would typically be 1 metre and the inter-mass space also 1 metre, the overall width of the machine thus being 3 metres. The tractor 10 could weigh 8 tons. From operational experience it is found necessary to have the inter-mass space equal to or less than the width of each mass so that a second compaction pass could position a mass to cover the gap left on the first pass. Thus the width of each mass defines also the gap between masses and this requirement restricts the width 12 (Figures 2

and 3) to a narrow wheelbase.

The foregoing dimensional data is given in order to establish a conceptual grasp of the magnitude of the forces and stresses to which the machine components are subjected when an impact roller is in operation.

In discussing the operation of impact rollers, it is helpful to consider the two modes of operation of such machines. Firstly there is the "Carry" mode, in which the compactor masses are lifted clear of the ground by operation of the lift cylinders 8 (Figure 1) so that the impact roller may be driven or transported without applying impact blows to the ground. In this mode, the weight of the compactor masses is taken by the ground engaging wheels 7 Figure 1. Secondly, there is the "Compacting" mode in which the cylinders 8 are retracted and the compactor masses rest upon the ground such that when the carriage is towed, the compactor masses rotate and apply impact blows to the ground.

Carry mode problems of an impact roller of conventional type will first be described. Referring again to Figures 1, 2 and 3, stability against overturning about the roll axis is dependent entirely on the carriage wheels 7. It will be appreciated that as previously described the carriage is connected though pivots to the drop-link 5 and the drag-link 4 which is finally attached to the axle assembly 3, no relative roll axis movement being permitted between the carriage and the axle. In Figure 7 the vector lines 16 represent the forces through the centres of gravity of the masses, spaced apart 2m in a typical machine. The vector lines 17 represent reaction forces from the tyres, these typically being spaced apart by 0.5m. The adverse ratio of 2.0m to 0.5m arises from the design constraint of having to place the

wheels between the pair of masses. The effect of this adverse width ratio between the compactor masses and the ground engaging wheels when the impact roller traverses the typically uneven ground surface conditions encountered on construction sites is illustrated in Figure 7. A wheel 7, on entering a small surface depression 18, causes a magnified overturning roll axis movement of the compactor masses. As a result, one of the compactor masses may strike the ground at a point 19. This is most undesirable as it may result in damage to made or partly made surfaces, or to the machine itself. In practice it is found that the elasticity of the rubber tyres used on wheels 7 exacerbates these undesirable roll axis movements, and often results in oscillations of the compactor masses from side to side about the roll axis. The universal coupling 9 in Figure 1 between the carriage and the tractor offers no resistance to the roll axis motion of the impact roller with its attached compactor masses.

A practical operational requirement in the use of impact rollers is however that the machine be deployed in the carry mode from one work site to another often several kilometres distant at a reasonable speed, desirably in excess of 20 km/h. The undesirable roll axis movements described above however limit the practical carry mode speed to less than 5 km/h for existing dual mass machines.

Roll axis stability in the carry mode needs to be achieved for a further reason. Consider a machine operating in the compaction mode, to achieve a number of compaction passes, typically twenty. An impact roller compacts in one direction only, therefore at the end of each compaction pass the machine must be turned about to make the return pass. The turning movement requires a space of three or four times the overall machine width,

which space is often not available. Furthermore the soil surface in the turning area becomes disturbed and uneven, causing the impact roller to slow down with consequent operator discomfort, aggravated wear on the machine and loss of productivity. The area in which an impact roller turns is not generally compacted to specification due to the unevenness of coverage. If instead of turning the machine at the end of each pass, it were possible to jack the masses up into the carry mode and reverse rapidly to the start position for the next pass, a great increase in machine operational efficiency would be achieved. Indeed on some narrow embankment projects such as railway embankments and water retaining bunds, work takes place above the level of the surrounding countryside so that it is not practical to use a compaction machine which requires a space of several times its own width to turn about.

Clearly there is a requirement to provide adequate stability of the masses in the carry mode, and accurate steering at speed in reverse. In the conventional dual-mass impact roller configuration wherein an impact roller is hitched to a four-wheeled towing tractor by means of a universal coupling 9 as shown in Figures 1, 2 and 3, the process of reversing is equivalent to reversing a trailer: reversing speed is thus slow for fear of jack-knifing and steering is inaccurate.

The deficiencies of existing dual mass designs when operating in the carry mode have been noted in the foregoing. One of the objectives of the present invention is to enable the pair of masses to be supported in the carry mode in a sufficiently stable manner that reasonably rapid transport over unmade construction terrain or bituminous surfaced roads is possible, in either a forward or reverse direction with little or no dangerous side to side sway, or

damaging ground contact.

In the compacting mode the machine typified by Figures 1, 2 and 3 also has deficiencies which will now be described. In such a design in which the axle 3 and the carriage 1 are coupled to each other with no freedom of movement relative to one another about the roll axis, high stresses due to shock loads are induced in the carriage and linkage components. This phenomenon is explained by the reference to Figures 5 and 6. When the machine is in operation, the masses 2 rotate in synchronism. Were the ground surface to be perfectly flat, they would strike the ground simultaneously and the connecting axle 3 would remain parallel to the ground, rising with the masses as the rolling motion continues. In reality, however, the ground surface is disturbed and has rises and dips at random. Figure 5 shows the near-side mass of a pair striking a rise, illustrated by a boulder 13. Figures 5 and 6 show that the centre of the mass striking the boulder begins to rise, the axle 3 consequently undergoing a roll axis rotation in the direction of the arrow 15. In Figures 5 and 6 the rare event of striking a boulder 13 is used to explain roll axis displacement. During normal operation in the compacting mode however, differential soil collapse between one side and the other under the impact blows can result similarly in varying degrees of roll axis displacement of the axle. It is quite usual for there to be a difference in settlement of 50mm between one side and the other, creating cyclical roll axis angular displacements such as that indicated by the numeral 15 in Figure 6. The time interval in which one side settles relative to the other by an arbitrary figure of 50mm can be approximately 30 milliseconds.

The roll axis angular displacement described above and illustrated by the

arrow 15 in Figure 6 carries with it (referring to Figures 1 and 3) the drag-link 4, the drop-link 5 and the carriage 1. It will be appreciated that large roll axis torque forces and consequent stresses result from transferring this roll axis displacement to the heavy linkage and carriage components occur within the short time interval of typically 30 milliseconds. These short duration roll axis displacements (of the order of 30 ms) are referred to as "shock roll displacements" and the stresses arising therefrom are referred to as shock roll stresses.

It will be evident from the foregoing that the degree of stressing of the linkage system connecting the axle 3 to the carriage 1 increases as the mass of the carriage increases. Any change in design of the carriage which makes it wider or heavier must therefore also deal with the matter of shock roll stresses.

The significance of shock stresses requires explanation in relation to the operation of impact rollers. A typical five-sided impact roller strikes the road surface at approximately 2 blows per second, which in 10 hours of machine operation gives 2 x 60 x 60 x 10 = 72 000 blows. In an average month's work of 100 hours 720 000 blows are produced. With each blow giving rise to stress in the machine parts there would be 720 000 stress pulses each month. It is well understood by machine designers that stress repetition of this order of magnitude could produce fatigue failure in the metal components. The most important factor in determining the number of stress repetitions which the metal can accommodate before failure occurs is the peak value of the stress pulses. Thus to ensure an acceptable working life for the machine, typically several years, it is important to provide means to reduce the peak values of stress. The most severe peak values of stress are

produced by the shock roll displacements acting against the inertia of heavy carriage components, as previously described. This gives rise to the necessity of providing a central pivot on the axle, by which means shock displacements of typically 2 to 4 degrees of rotation are accommodated and transferred to the heavy carriage components over a sufficiently long time interval to attenuate other wise damaging stresses.

Roll axis displacements of the masses and carriage combination in relation to the towing tractor also occur due to side slope distortion in the ground surface, but these are comparatively slow movements, which are accommodated by the pivot 9 between the carriage 1 and the tractor 10 in Figure 1. These slower movements, which do not involve shock loadings, are referred to as "slow roll displacements".

In the foregoing paragraphs, problems have been described which are detrimental to effective and safe operation of impact rollers in their present form: firstly, that of roll axis instability in the carry mode, and secondly, that of shock roll displacements in the compacting mode, from which severe stresses arise. It is therefore the objective of this invention to provide means whereby an impact rolling machine can travel safely and steer accurately in both forward and reverse directions at speeds appropriate to the deployment of construction plant on sites, for example approximately 20km/h, carrying the pair of compactor masses on its wheeled chassis. At the same time the invention seeks to provide means to alleviate high stresses which result from shock roll axis displacements when operating in the compacting mode.

SUMMARY OF THE INVENTION

According to the invention there is provided a self-propelled impact compaction machine which comprises a chassis, a prime mover on the chassis, ground engaging wheels on the chassis at least some of which are driven by the prime mover, and one or more impact compactor masses connected resiliently to the chassis, the impact compactor mass or masses defining a compaction track width measured from one lateral extremity of the mass or masses to the opposite lateral extremity of the mass or masses, and the ground engaging wheels of the chassis defining a maximum wheel track width which is not substantially different in magnitude to the compaction track width so that the chassis enjoys substantial roll axis stability.

The machine preferably includes a pair of spaced apart impact compactor masses supported on a common axle connected resiliently to the chassis, with one or more ground engaging wheels mounted to the chassis between the masses. In this arrangement, the maximum wheel track width defined by the ground engaging wheels of the chassis is preferably approximately equal to or only slightly less than the compaction track width defined by the laterally outer extremities of the two masses. The machine in such versions of the invention will also include lifting means, acting between the chassis and the compactor masses, to elevate the compactor masses above the ground for transportation thereof.

In one form of the invention, preferably of the dual mass type set forth above, the chassis of the machine is provided by a single, unitary, rigid structure. In this case, the driven, ground-engaging wheels may be carried

by a single drive axle powered by the prime mover. In this form of the invention, it is preferred that the axle supporting the compactor mass or masses has at least a small amount of roll axis freedom relative to the chassis. Damping means are preferably provided to damp roll axis movements of the axle relative to the chassis.

In another, steerable form of the invention, once again preferably of the dual mass type set forth above, the chassis of the machine has fore and aft parts articulated to one another at an upright yaw axis. There is preferably also a roll axis pivot between the fore and aft parts. Damping means are preferably provided to damp relative roll axis movements of the aft part of the chassis relative to the fore part at least over a predetermined range of pivotal movements about the roll axis. The machine may, in this case, include a breakaway mechanism allowing undamped pivotal movement of the aft part relative to the fore in the event of roll axis movement between the parts exceeding a predetermined limit.

In either form of the invention, the roll axis damping means may be adjustable to vary the resistance which is offered to relative roll axis movements. In a carry or transportation mode, with the impact compactor masses lifted clear of the ground and hence with a greater propensity to overturning, the damping means will typically provide greater resistance to roll axis movements than in an operative or compaction mode with the impact compactor masses in contact with the ground. In some cases the damping means, preferably provided by hydraulic dampers, may be varied manually while in other cases, there may be automatic control dependent on the operation of steering, braking or other systems of the machine.

In the articulated form of the machine, slow roll displacements of the aft part of the chassis relative to the fore part of the chassis are accommodated by the roll axis pivot. In this case, the machine may also include a secondary roll axis pivot, which is designed to take account of shock roll movements and which is typically located at the connection between the axle and a resilient linkage which connects the axle to the chassis. The pivot in this case is preferably also damped. In one specific embodiment, dampers of rubber or other resilient material provide both the pivoting and damping effects.

BRIEF DESCRIPTION OF THE DRAWINGS

Figures 1 to 7 of the accompanying drawings illustrate a conventional dual-mass impact roller. In these Figures:

Figure 1 shows a side elevation of a conventional towed impact roller, with one compactor mass removed to show details of the carriage and linkage system;

Figure 2 shows a rear elevation of the impact roller seen in Figure 1;

Figure 3 shows a plan view of the impact roller seen in Figure 1;

Figure 4 shows a diagrammatic representation of the measurements required for the determination of the

drop height of an impact roller;

Figure 5 shows a diagrammatic side elevation of a pair of compactor masses striking an uneven site surface;

Figure 6 shows a diagrammatic rear elevation of a pair of compactor masses striking an uneven site surface, and indicates the direction of angular acceleration of the compactor masses and connecting axle assembly; and

Figure 7 shows a diagrammatic representation of the weight distribution of a pair of compactor masses in the carrying position supported by a pair of narrowly spaced wheels, and illustrates instability when travelling over uneven ground.

Figures 8 to 23 illustrate embodiments of the invention, by way of example only. In these Figures:

Figure 8 shows a diagrammatic plan view of a rigid T- shaped chassis frame of an impact roller with drive-steer axle, in accordance with the principles of the invention;

Figure 9 shows a diagrammatic plan view of a T-shaped chassis frame with the articulated system of steering;

Figure 10 shows a partly sectioned side elevation of an impact roller according to the invention, one compactor mass

being removed so as to expose the working mechanism and chassis details;

Figure 11 shows a plan view of the machine seen in Figure 10, with certain details omitted in the interests of clarity;

Figure 12 shows a rear elevation of the machine seen in Figure 10, with certain details omitted in the interests of clarity;

Figure 13 shows a partly sectioned rear view of the roll axis pivot seen in Figure 10;

Figure 14 shows a sectional view of the mechanism seen in Figure 13;

Figure 15 shows a partly sectioned side elevation of an impact roller according to an alternative embodiment of the invention;

Figure 16 shows a plan view of the impact roller seen in Figure 15;

Figure 17 shows a sectioned view, on an enlarged scale, of the pivot and restraining mechanism of the impact roller shown in Figures 15 and 16;

Figure 18 shows an elevation and part sectioned view of the

mechanism of Figure 17;

Figure 19 shows, to an enlarged scale, details of the drag-link and elasticised shock roll axis stress attenuation and orientation mechanism seen in Figure 15;

Figure 20 shows an end elevation of the mechanism seen in Figure 19;

Figure 21 shows a plan view of the axle tube of Figure 19 indicating the positioning of the elastic elements;

Figure 22 shows in side elevation a further alternative embodiment of the invention; and

Figure 23 shows, in plan view, a detail of the pivot mechanism seen in Figure 22.

DESCRIPTION OF EMBODIMENTS

Reference is made firstly to Figures 8 and 9 which illustrate the underlying principle of the invention. These Figures show an impact roller chassis frame 20 which is T-shaped in plan view. Steerable wheels 21 are mounted on an axle 22 located beneath the transverse part 20A of the chassis frame. The axle 22 may be powered by a suitable motor and transmission located on the transverse part 20A, which can additionally provide seating and controls for an operator. As an alternative, the chassis frame can be adapted for flexible

coupling to a towing tractor.

A further wheel 23 is mounted in a freely rotatable manner at the end of the web 20B of the T-shaped chassis frame. The web of the chassis frame is located between compactor masses 24. It will be noted that the compactor masses are quite closely spaced, this being permitted by the narrowness of the web 20B of the chassis frame. The masses 24 are connected to one another for substantially synchronous rotation by a common axle 25. Other components (not illustrated) of the linkage between the chassis frame and the compactor masses may be the same as in the conventional linkage previously described with reference to Figures 1 to 3 of the drawings.

The single wheels 21 and 23 can, in other embodiments of the invention, be replaced by groups of wheels at the relevant locations.

Figure 9 illustrates a modified chassis frame in which the transverse part 20A of the chassis frame is pivoted to the web 20B of the frame at a vertical axis 27. In this case, the axle on which the wheels 21 are mounted can be steered as well as driven. In a case where the axle is not driven, the steerable wheels will merely allow the impact roller to follow the direction of travel of the towing tractor. It may in such a case be necessary to weight the transverse chassis frame part 20A to ensure adequate stability.

During operation of the impact roller in the compaction mode, shock roll-axis displacements of the axle occur, as previously described. To relieve the stresses which these shock displacements would produce on the linkage and heavy chassis with integral tractor components 20A and 20B of Figures 8 and 9, it is desirable to allow for rotational movement of the axle 25 about

the fore and aft roll axis of the machine.

In practice, it is advantageous to arrange the necessary rotatable connection close to the axle 25, thereby limiting the mass which is accelerated about the roll axis. As is explained subsequently with reference to Figures 10 to 14, this can advantageously be achieved by pivotally mounting the axle 25 about its centre to the chassis and linkage components, with the provision of means to restrain movement about the yaw axis, i.e. the vertical axis.

It will be noted in Figures 8 and 9 that the wheels 21 are spaced apart by a considerable distance, in the illustrated cases approximately equal to the overall width of the machine as defined by the outer extremities of the compactor masses 24. This gives stability to the impact roller and reduces the chances of the impact roller overturning about the roll axis when compared to the conventional impact roller as described with reference to Figures 1 to 3.

Reference is now made to Figures 10 to 14 which illustrate a practical, self-powered and steerable embodiment of the invention. These Figures show a soil compaction machine 30 comprising a T-shaped, motorised chassis frame 31 supported upon a pair of driven ground-engaging wheels 32. The wheels 32 are located at either extremity of a transverse axle (not visible in the drawings) which is both driven and steerable and which is carried by the transverse part of the T-shaped chassis frame.

The chassis frame 31 is also supported by tandem pairs of wheels 33 located towards the end of the web part 34 of the chassis frame. As in Figures 8 and 9, the web part of the chassis frame 31 is of narrow design with the

clear space between the compactor masses being sufficient only to accommodate the web part and the wheels. Also as in Figures 8 and 9, the overall transverse dimension of the driven, steerable axle plus the wheels 32 which it carries is approximately equal to the overall width defined by the compactor masses 35. Thus good anti-roll stability is once again achieved.

Each compactor mass 35 is a three sided roller with three peripherally spaced salient points 36. Each salient point is followed by a re-entrant portion 37, and each re-entrant portion 37 is followed by a compacting face 38. The compactor masses are mounted fast on a common axle 39 in an axle housing 40 so as to rotate in unison when the impact roller moves over the soil surface.

Pinned at a pivot point 41 to a rear bulkhead 42 of the chassis frame 31 is a generally horizontal drag-link 44 and a vertical link 45. The drag link 44 and the vertical link 45 are able to rotate about the pivot point 41 independently of one another. An upper extension of the vertical link 45 is pinned at a pivot point 46 to the piston rod of an hydraulic cylinder 47, acting as an hydraulic spring.

To raise the masses to the carry mode, a jack 69 is extended, bringing a plate 71, which is hinged at 72, into contact with the underside of the drag-link 44. The drag-link 44 is thereby raised, carrying the axle assembly and compactor masses with it.

The base of the hydraulic spring 47, which operates in a generally horizontal direction, is pinned to the bulkhead 42 at a pivot point 48. The hydraulic spring 47 has an hydraulically actuated ram rod which is connected to a gas

charged accumulator (not shown) so as to provide required load deflection characteristics. The vertical link 45 is provided with bumper plates 49 adjacent to the central pivot point 41, and rubber bumpers 50 are fixed to the bulkhead 42 opposite the bumper plates 49, so as to provide end-of-travel limit stops for the vertical link 45.

The rearmost extremity of the drag link 44 is pinned to a drop link 51 at a pivot point 52. The upper part of the drop link 51 is pinned at a pivot pint 53 to the rearmost extremity of a spacer bar 54, and the forward extremity of the spacer bar 54 is pinned at a pivot point 55 to the vertical link 45.

It will be noted that the pivot point 55 is located between the pivot points 41 and 46. The lower part of the drop link 51 is in the form of a large clevis which straddles the axle housing 40 and which is connected pivotally to the housing 40 at a pivot assembly 56.

The drag link 44, vertical link 45, spacer bar 54 and drop link 51, along with their associated pivots and the hydraulic spring 47, together constitute the linkage system. This linkage system constrains the axle housing 40 to move forward and backward relative to the bulkhead 42 without permitting yaw axis freedom, and allows the axle housing 40 to move up and down with freedom to rotate about the roll axis, i.e. about the pivot assembly 56.

It will be appreciated that the connection between the chassis frame 31 and the compactor masses 35 is resilient in nature. In operation the motor and transmission mounted forwardly on the chassis frame 31 operate on the driven wheels 32 to move the machine forward in the direction of the arrow 57 in Figure 10, pulling the compactor masses 35 along by means of the

resilient linkage system. Aside from slight asynchronism permitted by torsional resilience of the common axle 39, the compactor masses, in contact with the ground, rotate in unison with one another.

During such rotation of each compactor mass it undergoes a repetitive sequence in which the mass rises up on a salient point 36 then, once an over-centre condition is reached, drops downwardly and forwardly for the following compacting face 38 to impact against the ground surface.

Should one of the compactor masses 35 strike a raised area of soil, such as the zone 13 exemplified in Figures 5 and 6, the pivot assembly 56 allows the combined assembly of the two compactor masses 35, the common axle 39 and the axle housing 40 to rotate about the roll axis of the impact roller. This avoids excessive stress on the linkage system. In Figure 6, the numeral 14 indicates the average ground surface, and as explained previously, the arrow 15 denotes the roll axis movement of the pair of compactor masses as one of them strikes the raised zone or obstruction 13.

Figures 13 and 14 show the details of the pivot assembly 56. The tubular axle housing 40 has flat plates 59 let into its sides towards the front and back. The lower clevis of the drop link 51 straddles the flat areas defined by the plates 59 and is connected pivotally to the plates by means of pivot pins 60 and bushes 61. Spaced apart laterally at the extremities of the drop-link clevis are four restraining pads 62 which act against steel rubbing plates 58 to prevent movement of the axle housing assembly about the yaw axis i.e. about a vertical axis, relative to the chassis frame 31. The steel rubbing plates 58 are welded fast to the plates 59. The restraining pads 62, which are constructed from nylon material, are adjustable to compensate for wear

by means of adjuster screws 63.

To control the rate of roll axis rotation of the axle housing 40 about the axis of the pivot assembly 56, and to control the degree of rotation of same, two hydraulic damper cylinders 64 are provided. The piston rods of the cylinders 64 are pinned at pivot points 67 to the drag-link 44 and the cylinders themselves are pinned at slotted pivot points 68 to the axle housing 40.

The two cylinders 64 are connected to a gas-charged hydraulic accumulator 66 via adjustable restriction orifices 70 that allow restricted flow of hydraulic fluid from the cylinders to the accumulator.

Connected in parallel to each adjustable orifice 70 is a non-return valve 65 that allows free flow from the accumulator 66 to the cylinders 64. In a static position, with the axle housing 40 generally parallel to the drag-link 44, the piston rods of the damper cylinders 64 are both in their fully extended position due to the pressure in the accumulator. In these positions, the pistons 75 are situated inside the cylinders at their uppermost limits, and the lower pivot pins are at the lower limit of their travel within the slots at the pivot points 68.

If a roll axis displacement of the axle housing 40 relative to the orientation of the draglink 44 occurs, the hydraulic components function as follows. It is assumed that the right hand side of the axle housing 40, as viewed in Figure 13, is displaced upwardly. Since the cylinder stop pin is already seated at the bottom of the slot 68, the cylinder 64 commences upward movement, being carried by the movement of the axle. Fluid within the

cylinder is forced through a hole drilled in the centre of the piston crown 75. Because there is no escape for fluid around the piston crown 75 as a result of the presence of a seal 75a, fluid flows through a port 75b. The non-return valve 65 closes so that fluid flows through the orifice 70, thereby controlling the rate of roll axis movement of the axle housing 40 about the pivot 56.

Considering now the left hand cylinder 64, the cylinder stop pin remains stationary while the lug on the axle housing 40 with its slot 68 moves downwardly until the upper extremity of the slot engages the stop pin, thereby defining the limit of permitted rotational movement of the axle. Simultaneously, the limit of travel of the right hand piston is reached. Fluid displaced from the right hand cylinder cannot be accommodated in the left hand cylinder as it is already full. The fluid therefore passes into the accumulator 66 which sustains a preset fluid pressure. As soon as the disturbing torque on the axle housing 40 is removed, fluid under pressure flows without restriction through the non-return valve 65 to produce a force between the right hand piston 75 and cylinder 64 which tends to restore parallelism between the draglink and the axle housing 40.

Sustained fluid pressure from the accumulator 66 operates with both the left and right hand cylinders 64 to provide a self-centring torque between the axle housing 40 and the draglink 44. The magnitude of this self-centring torque may be adjusted by operation of the fluid control valve 73 to either drain fluid from the accumulator 66 to the reservoir 74, thereby reducing the fluid pressure, or to charge fluid from the pump 80 to the accumulator, thereby increasing the fluid pressure.

By means of the self-centring mechanism as described above, the machine

may be operated in the carry mode with the pair of masses restrained against random roll axis movements. In this way it is possible to achieve safe transportation, at adequate speed, between work sites.

Figures 10 to 14 thus show an embodiment of the invention based on the integral carriage and steering system shown in Figure 8. An application of the invention to the carriage and steering system as shown in Figure 9 will now be described.

Figure 9 shows a carriage similar to that of Figure 8 but with steering of the drive wheels accomplished by means of a vertical yaw axis pivot 27. This is therefore an articulated carriage system. Articulated carriage systems are widely used in the construction industry for equipment such as front end loaders, soil compactors and the like. In addition to the vertical pivot it is standard practice in such conventional machines to accommodate slow-roll movements due to ground unevenness by providing a roll axis pivot generally allowing limited angular movement. In these conventional articulated designs two sections of the machine each having stability against overturning during normal operation are coupled together at the combined yaw and roll axis pivot point. In the adaptation of the articulated steering system to impact rollers however, the part of the carriage indicated by numeral 20B in Figure 9 has intrinsic stability against overturning only when both masses are in contact with the ground, i.e. when operating in the compacting mode. When the masses are jacked up to be carried on the narrow wheel or wheels 23 of Figure 9, located between the masses, the carriage part 20B is unstable about the roll axis. Therefore the roll axis pivot of an articulated type of carriage adapted for dual mass impact rollers requires to have a restraining torque applied to enable the stable wide wheel

base part 20A to offer the required resistance to overturning of the part 20B.

When the masses are in contact with the ground, i.e. in the compacting mode, another consideration related to the operating characteristics of impact compaction machines dictates the need for a roll axis restraining torque to be applied at the coupling between the carriage portions 20A and 20B of Figure 9.

Referring again to Figure 1, consider the masses 2 to be operating in contact with the ground. As the mass assembly passes over a salient point b in motion, it tends to accelerate forward and downward. In steady motion at operating speed this forward and downward movement is taken up by the action of the linkage components 4, 5 and 6 between the chassis frame and the compactor masses and axle assembly, and little or no undesirable acceleration forces are applied to the tractor 10. However at slow speeds and during stopping, starting and manoeuvring, load pulses are produced by the drop-link 5 striking against the end of travel stops a. The net result of such load pulses on the coupling 9 is that upward or downward forces are produced at this point. Consider the effect of such forces acting on the articulated tractor system shown in Figure 9. The forces acting at the pivot 27 when travelling in a straight line as drawn in Figure 9 do not produce any motion about the pivot 27, as the pivot does not allow any pitch axis rotation. However, when the steering is operated to bring the drive wheels into the position 20C shown in broken outline, the downward or upward force operating at the pivot point 27 has a leverage about the point c with a lever length d about the axis of the pivot. This force couple induces the "tractor", or forward chassis portion 20A, to execute a pitching motion about its wheels. Due to the geometry of the pivot system 27, such pitching

movement of the chassis portion 20A is accompanied by pitching and roll axis motion of carriage element 20B. Similar pitching action of the tractor element also results from torque reaction and braking forces generated by the action of the driven wheels 21 whenever the two chassis elements are not in an exactly straight line.

The foregoing analysis establishes the need for an articulated carriage system for an impact roller to provide the conventional vertical axis, or yaw, articulation pivot, and additionally a roll axis pivot to accommodate slow roll axis movement between the two parts 20A and 20B of Figure 9. However to allow part 20B of the machine (Figure 9) to have stability against overturning in the carry mode, the roll axis pivot requires a restraint against free roll axis movements. The roll axis restraint should however be such that the degree of restraint may be varied from light torque restraint when the machine is operating in the compaction mode to medium torque restraint when the machine is in the carry mode operating in a straight line, to high torque restraint when the machine is travelling at high speed or when manoeuvring in the carry mode. Conditions such as wheels or masses stuck in loose sand, muddy conditions, conditions when the traction wheels are at full torque forwards and backwards, or conditions in which the masses are raised to the carry mode in adverse travel conditions, or any combination of these conditions, would require that the articulation of the roll axis pivot be locked in the vertical orientation to prevent over-toppling of the carriage part 20B or rotation of the part 20A about its wheel axis.

An embodiment of an articulated machine incorporating a roll axis restraining system is now described with reference to Figures 15 to 18.

Figure 15 shows an elevation with one mass removed, and partly sectioned, to show the construction of the machine parts. Figure 16 shows a plan view, partly sectioned, of the machine seen in Figure 15. The motorised chassis 80 of the articulated machine is coupled by a generally vertical pivot system 86 to a trailing carriage 81 including a carriage extension 84 which is sufficiently narrow to allow a set of carriage wheels 83 to be fitted in the space between the compactor masses 85. Drive wheels 82 are mounted to the tractor chassis 80. Steering of the machine is effected by operating steering jacks 87. Between the chassis 80 and the carriage 81 is a pivot housing 89 within which are located the roll axis and yaw axis pivots and control mechanisms.

A pivot shaft 88, which is flanged and secured to the carriage 81 by bolts 90 and which protrudes through a machined hole in the pivot housing 89, carries a pivot bearing 91 and a thrust bearing 92 which reacts against a thrust bearing 93 located in an annular groove in the mating face of the pivot housing 89. A flange 94 is secured to the pivot shaft to retain and pre-load the thrust bearing. The flange carries crank-pins 95 by means of which rotational torque is applied to the pivot shaft 88 by the operation of a pair of hydraulic rams 96. For clarity the internal mechanisms of the pivot housing 89 are shown (with some detail omitted) in Figures 15 and 16. Detail is shown to a larger scale in Figures 17 and 18 which also illustrate a break-away safety system.

Figure 17 shows a detail of the pivot system connecting the carriage 81 to the pivot housing 89. Numerals correspond with those in Figures 15 and 16, but additional numerals are incorporated to show detailed features. Figure 18 shows cross-sections at the lines A-A and B-B in Figure 17.

The flange 94 in Figure 17 is fixed against rotation by a key 122 engaging the pivot shaft 88, and retains the pivot and thrust bearings 91 and 92. Pre-load on the thrust bearings 92 and 93 is achieved by fastening screws 124 passing through a retainer washer 120 into the pivot shaft 88. A crank flange 121 is enabled, by bearings 123, to rotate relative to the flange 94, but during normal operation of the machine is constrained to follow the random clockwise and counter-clockwise movements of the flange 94 by location of a pre-loaded, vee-shaped pin 126 in a corresponding detent. The pin 126 includes an hydraulic cylinder, of which the vee-shaped element is the piston rod. Note that these random movements occur when the chassis 81 moves about the roll axis relative to the pivot housing 89, but during normal operation stay within a range of about 15 degrees on either side of the vertical.

The crank pins 95 are pivotally attached to the piston rods 127 of the cylinders 96. In each case, operation of hydraulic pressure at a port 128 applies a force to the piston rod 127 equal to the cross sectional area of the rod piston multiplied by the hydraulic pressure. The cylinder 96 is thereby extended to cause a hinge plate 129 to bear against a stop block 130. The lengths of the piston rods 127 are selected such that with both pistons extended to the extremity of their travel, the carriage 81 is in correct vertical alignment with the pivot housing 89, and the pin 126 is seated in its corresponding detent in the flange 94.

Assume that the carriage 81 rotates about the roll axis relative to the pivot housing 89. The flange 94 carries with it the crank flange 121, which in turn forces (say) the left-hand piston rod 127 into the cylinder 96 with a damping action controlled by the setting of a variable restrictor valve 131.

On the other, right hand side, the crank pin 95 draws the piston rod and cylinder downwardly, the top end of the cylinder carrying with it, without restraint, the right hand hinge plate 129. The hydraulic pressure in the left hand cylinder will tend to cause the cylinder assembly to extend to its full extension, and in this manner the carriage 81 and the pivot housing 89 are biased by a pre-set torque to be in alignment over the arc of allowed travel, in this case 30°. The pre-set torque may be adjusted by varying the hydraulic pressure applied to the system, and the damping may be adjusted by varying the setting of the restrictor valve 131.

Now consider that an event such as a landslip overturns the carriage 81 to an angle greater than the arc of allowed travel, which is 30° in the case of the mechanism shown in Figures 17 and 18, or 15° on either side of vertical. Due to its vee-shape, the pin 126 overcomes the pre-load placed upon it by hydraulic pressure applied at a port 132 and escapes from its detent. Rotation of the flange 94 can now continue independently of the crank-flange. When normal orientation is restored, the pin 126 is brought back into alignment with the detent and can slip back into its normal position in the detent under the effects of the hydraulic pressure supplied through the port 132. The hydraulic pressure at the port 132 can be varied in order to adjust the pre-load upon the pin 126, and a hydraulic pressure relief valve may be provided in this circuit to vent fluid to tank as the pin 126 moves into its cylinder or the excess fluid may be accommodated in a pressurised hydraulic accumulator.

An alternative to the hydraulic activation of the pin 126 would be to substitute a shear pin, but this is less desirable due to the difficulty of replacement in the field should a shear occur.

To transfer the tractive force from the carriage 81 to the axle 106, a linkage system is provided consisting of a drag-link 98, a drop-link 99 and a traction spring 97. The traction spring 97 is hydraulically operated to provide a nearly constant traction force whatever the amount of piston rod extension. At each extremity of the drop-link 99, bumper plates 101 are provided to operate against travel limiting bumpers 100, preferably constructed of hard rubber.

A pivot 102 provides pitch axis rotation of the drop-link 99 relative to the carriage 81, and a pivot 103 provides pitch axis rotation of the drag-link 98 relative to the drop-link 99.

With this linkage system, the extremity 104 of the drag-link 98 has freedom to move up, down, fore and aft, but there is no freedom of roll axis angular displacement relative to the carriage 81. The consequences of this are now described.

If the end 104 of the drag-link 98 were rigidly fixed to the axle housing 105 then any roll axis displacement of the axle housing 105 would induce the carriage 81 to follow the roll axis displacement and rotate about the pivot shaft 88.

Because the carriage 81 is heavy, with a large moment of inertia about the roll axis, its rotation lags in time relative to the rotation of the axle. Therefore with shock roll displacements there would be twisting of the drag-link 98 to absorb the lost motion between the axle housing 105 and the carriage 81.

In the embodiment of the invention shown in Figures 10 to 14, the same analysis applied, but in that case the drive/steer type of carriage 31 has a much larger mass and moment of inertia than the carriage portion 81 of Fig. 15, because it includes the engine, transmission and drive wheels. In the embodiment of Figures 10 to 14, the pivot 56 allows the axle 40 sufficient displacement relative to the carriage 31 to enable its rotational movement to be controlled by damping cylinders 64 so as to prevent excessive stress in carriage and linkage components.

In the embodiment of Figures 15 to 18, because the carriage 81 would in a practical design have a rotational moment of inertia less than the carriage 31 of Figure 10, typically by a factor of between two and five, it becomes possible to provide for differential movement between the axle housing 105 and the end of the drag-link 104 by way rubber elements 107 and 108 arranged in a particular manner as described below.

Figure 19 shows partly a side elevation and partly a cross-section at the longitudinal centreline of the machine. Figure 20 shows an end elevation and Figure 21 a plan view of the axle housing 105 with the drag-link removed to show the position and orientation of the rubber elements 107 and 108. These rubber elements, in the form of rubber pads, are able to resist compressive loads on their large area dimension with only slight deflection, typically of the order of 10% of the pad thickness. Their lateral dimension is designed to allow for shear deflections of typically 50% of the pad thickness.

To provide an elastic centre pivot for roll axis movement, rubber pads 108 are mounted on plate members 109 within the drag-link structure to act upon

a torque member 110 which is integral with the axle housing 105. Figure 21 shows how the compression pads 108 and shear pads 107 are mounted in pairs on either side of the axle housing.

When the lift jack 111 is operated, a lifting force is applied by the plates 109, through the compression pads 108, to lift the axle 105 along with the attached pair of compacting masses. Traction and stopping forces are resisted by the shear pads 107 acting in compression, any shear movement is simultaneously accommodated. Roll axis displacement between the axle housing 105 and the drag-link 104 deforms the pads 107 in shear until further movement is restrained by the axle 105 housing abutting a movement-limiting bumper 112. Roll axis displacement is not substantially restrained by the compression pads 108 because they merely deform to accommodate the movement.

For ease of assembly between the drag-link 104 and the axle housing 105, provision is made for bonding the pads 107 and 108 to the axle housing structure. The drag-link 104 is assembled over the axle housing to restrain the pads 108, with a clearance between the shear pads 107 and opposing compressor plates 113. Screws 115 are then tightened to pre-load the shear pads 107, and screws 116 are tightened to pre-load the compression pads 108.

Conventional articulated construction machinery utilizes a vertical axis pivot system with limited angular freedom of movement about the roll axis. This is achieved by a short top link which is free to swing to either side over a short arc. For reasons given earlier this system cannot safely be applied to impact rollers without biasing the yaw pivot to remain vertical.

In Figures 22 and 23 the vertical or yaw articulation axis is between the top and bottom ball joints 135. A top link 136 of the yaw pivot system is also pivoted at the carriage end by means of a ball joint 137. The link 136 can be constrained to be in line with the longitudinal axis of the carriage by applying hydraulic pressure to rams 138, in which case there would be no roll axis freedom between the carriage 81 and the motorised chassis 80. By reducing the hydraulic pressure on the two rams 138, sideways movement is permitted in response to a pre-set side force on the link 136 at the pivot point 137. The cylinders of the rams 138 are pivotally connected to a lever arm 139 so that when each ram-rod is fully extended, the lever abuts against a stop 140 which is adjustable for position by means of a screw 140A.

Steering of the articulated machine of Figures 22 and 23 is by operation of steering jacks 141.

The self-propelled impact compaction machines described above have a number of important features, including the following:

1. The driven wheels 21, 32, 82 are spaced a substantial distance apart from one another. The wheel track width, i.e. the lateral wheel to wheel dimension does not differ substantially from the compaction track width, i.e. the lateral dimension defined by the outer extremities of the compactor masses. This gives the machine considerable stability against overturning of the carriage which is on a narrow wheelbase between the masses. This principle is also applicable to single mass, as opposed to dual mass machines.

2. The roll axis pivot in each embodiment of the invention described

above allows for slow roll displacements and reduces the potentially damaging effects of roll axis movements while the damping and self-centring function associated with the roll axis pivot nevertheless give adequate stability. In the case of the articulated machine described with reference to Figures 15 to 21 there is provision for limited, damped roll axis movement at the axle connection between the axle housing and the drop link thereby taking account of potentially damaging shock roll displacements. The damping of the roll axis movements also reduces the possibility of potentially damaging oscillations.

3. In the embodiment of Figures 15 to 18, locking means, the pin 126 and its associated detent constitute a breakaway safety device which, in the event of excessive roll axis movement of the carriage, effectively disconnects the carriage from the traction part of the machine. Thus if the carriage experiences a roll axis movement of such magnitude that it tends to overturn, this movement is not transferred to the traction unit.

CLAIMS

1. A self-propelled impact compaction machine which comprises a chassis, a prime mover on the chassis, ground engaging wheels on the chassis at least some of which are driven by the prime mover, and one or more impact compactor masses connected resiliently to the chassis, the impact compactor mass or masses defining a compaction track width measured from one lateral extremity of the mass or masses to the opposite lateral extremity of the mass or masses, and the ground engaging wheels of the chassis defining a maximum wheel track width which is not substantially different in magnitude to the compaction track width so that the chassis enjoys substantial roll axis stability.

2. A self-propelled impact compaction machine according to claim 1 and comprising a pair of spaced apart impact compactor masses supported on a common axle connected resiliently to the chassis, with one or more ground engaging wheels mounted to the chassis between the masses.

3. A self-propelled impact compaction machine according to claim 2 wherein the maximum wheel track width defined by the ground engaging wheels of the chassis is approximately equal to, or only slightly less than, the compaction track width defined by the laterally outer extremities of the two masses.

4. A self-propelled impact compaction machine according to any either one of claims 2 or 3 and comprising lifting means, acting between the chassis and the compactor masses, to elevate the compactor masses above the ground for transportation thereof.

5. A self-propelled impact compaction machine according to any one of the preceding claims wherein the chassis of the machine is provided by a single, unitary, rigid structure.

6. A self-propelled impact compaction machine according to claim 5 wherein the driven, ground-engaging wheels are carried by a single drive axle powered by the prime mover.

7. A self-propelled impact compaction machine according to claim 6 wherein the axle supporting the compactor mass or masses has at least a small amount of roll axis freedom relative to the chassis.

8. A self-propelled impact compaction machine according to claim 7 and comprising damping means to damp roll axis movements of the axle relative to the chassis.

9. A self-propelled impact compaction machine according to any one of claims 1 to 4 wherein the chassis of the machine has fore and aft parts articulated to one another at an upright yaw axis.

10. A self-propelled impact compaction machine according to claim 9 and comprising a roll axis pivot between the fore and aft parts.

11. A self-propelled impact compaction machine according to claim 10 and comprising damping means to damp relative roll axis movements of the aft part of the chassis relative to the fore part at least over a predetermined range of pivotal movements about the roll axis pivot.

12. A self-propelled impact compaction machine according to claim 11 and comprising a breakaway mechanism allowing undamped pivotal movement of the aft part relative to the fore in the event of roll axis movement between the parts exceeding a predetermined limit.

13. A self-propelled impact compaction machine according to any one of claims 8, 11 or 12 wherein the roll axis damping means is adjustable to vary the resistance which is offered to relative roll axis movements.

14. A self-propelled impact compaction machine according to claim 13 wherein in a carry or transportation mode of the machine in which the impact compactor mass(es) are lifted clear of the ground the damping means provide greater resistance to roll axis movements than in an operative or compaction mode in which the impact compactor mass(es) is/are in contact with the ground.

15. A self-propelled impact compaction machine according to either one of claims 13 or 14 wherein the damping means are manually variable.

16. A self-propelled impact compaction machine according to either one of claims 13 or 14 and comprising means for automatically controlling the damping means in dependence on the operation of steering, braking or other systems of the machine.

17. A self-propelled impact compaction machine according to any one of claims 9 to 12, the machine comprising:

- a pair of spaced apart impact compactor masses supported on a common axle connected resiliently to the chassis, and

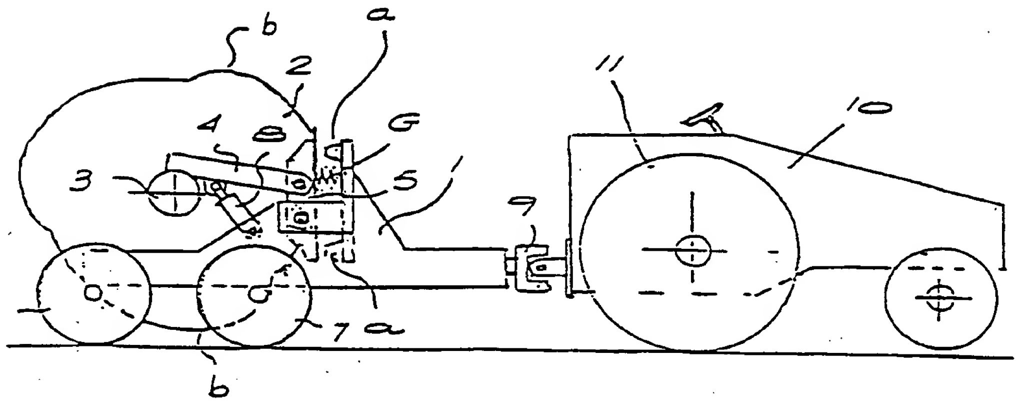
- a secondary roll axis pivot arranged to take account of shock roll movements of the axle.

18. A self-propelled impact compaction machine according to claim 17 wherein the secondary roll axis pivot is located at a connection between the axle and a resilient linkage which connects the axle to the chassis.

19. A self-propelled impact compaction machine according to claim 18 and comprising means for damping rolling movements about the secondary roll axis pivot.

20. A self-propelled impact compaction machine according to claim 19 wherein the damping means comprises dampers of rubber or other resilient material, the dampers providing both pivoting and damping effects.

FIG 1



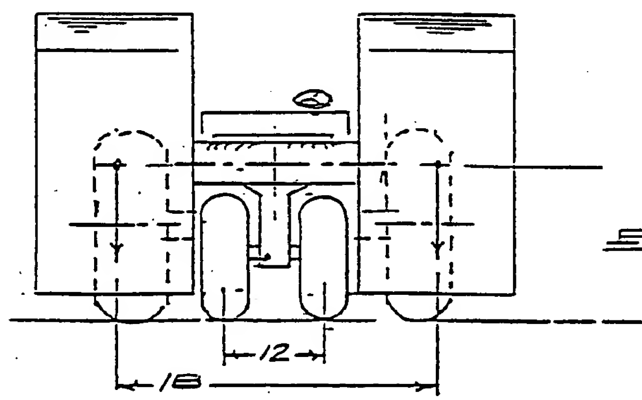


FIG 2

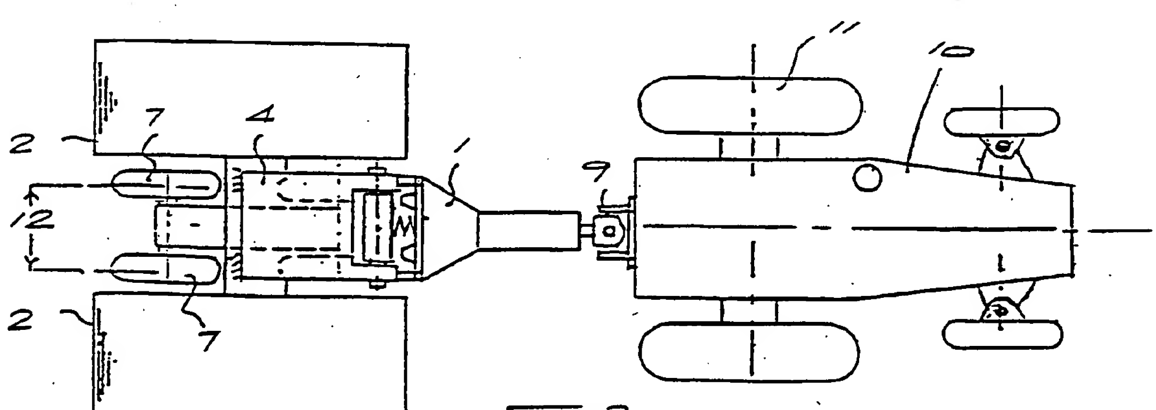


FIG 3

FIG 4

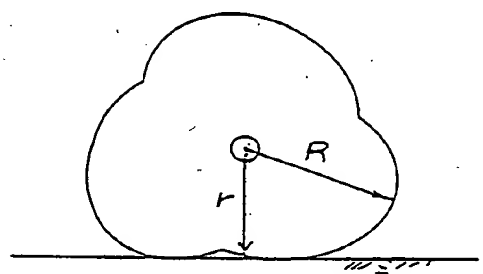


FIG 5

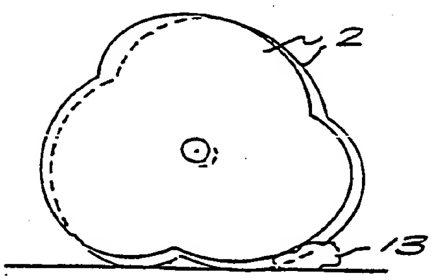
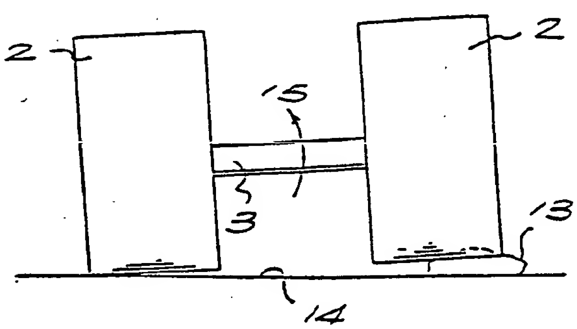


FIG 6



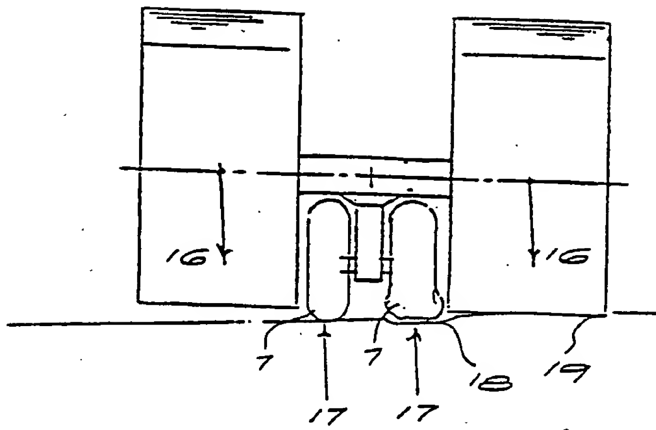


FIG 7

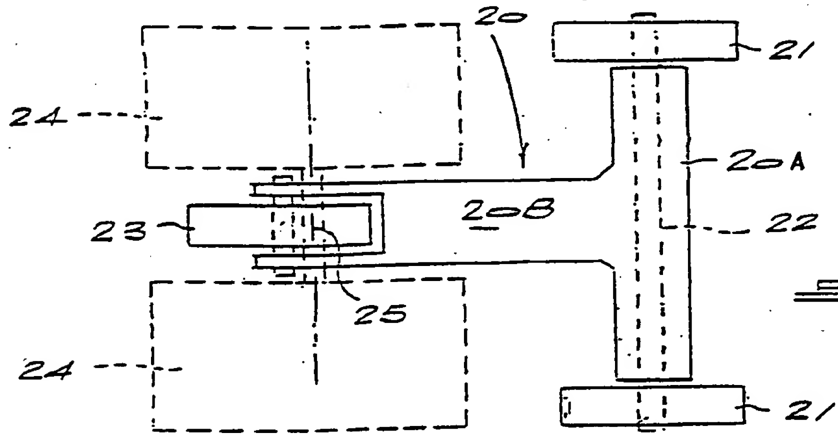
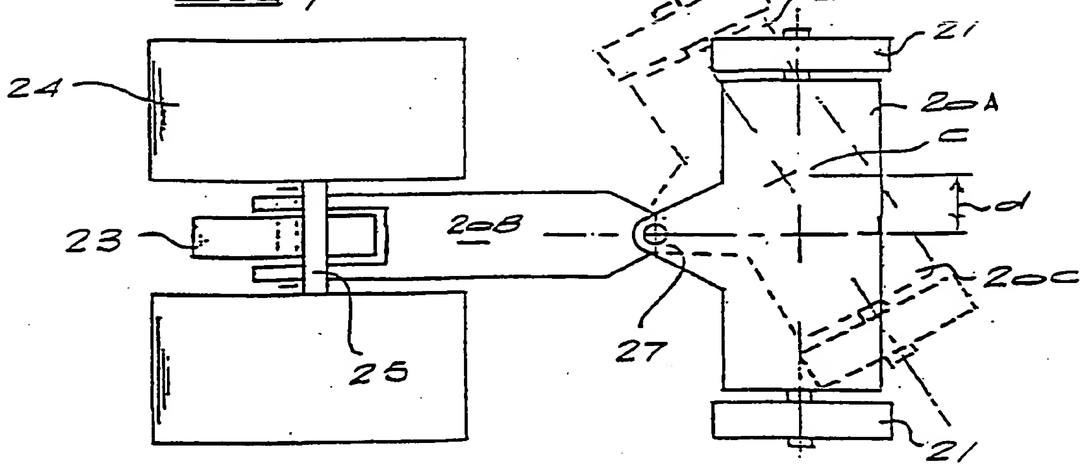


FIG 8

FIG 9



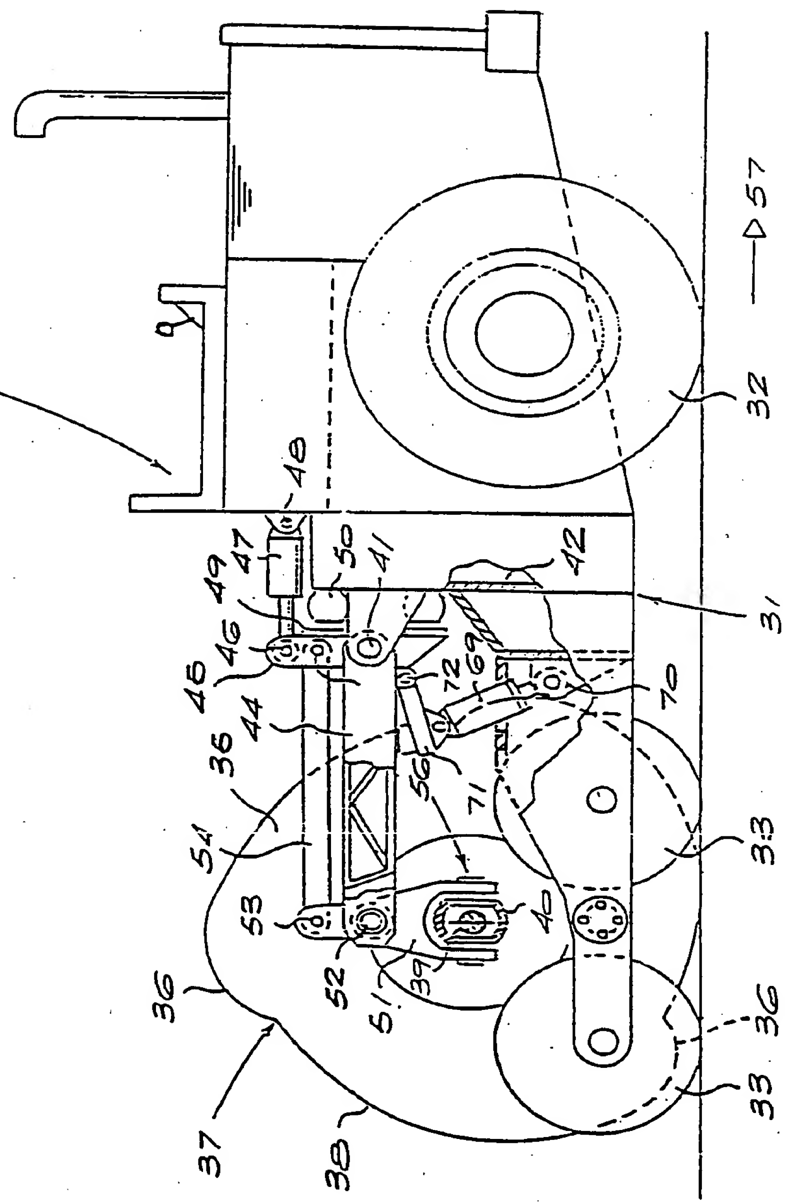

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FIG 10

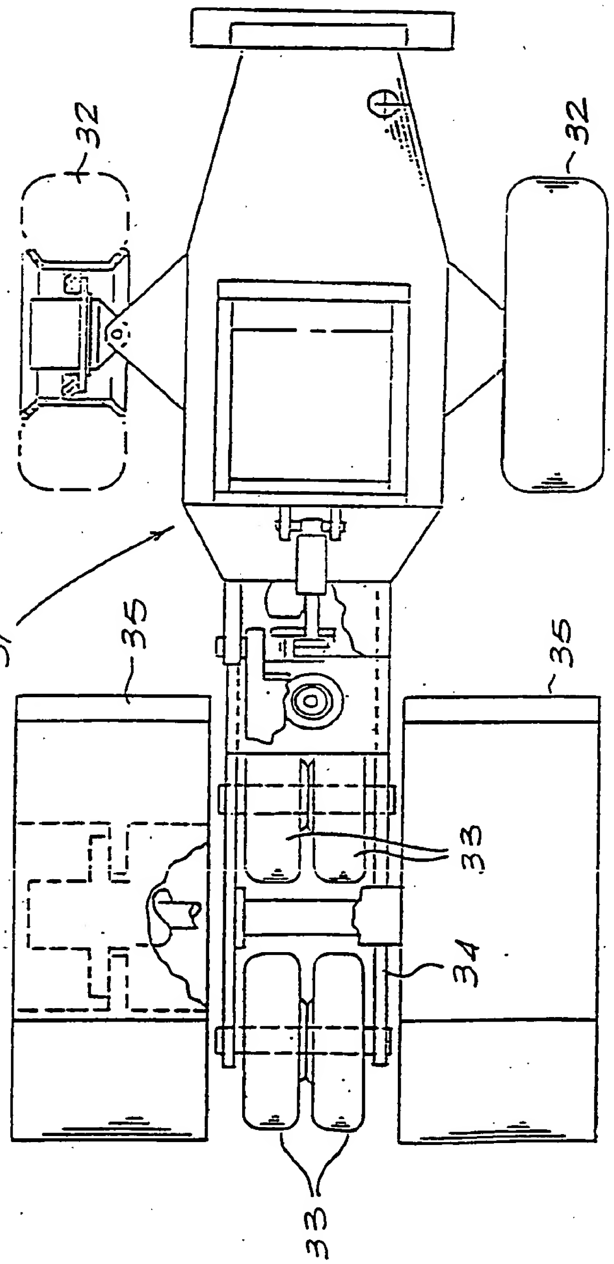
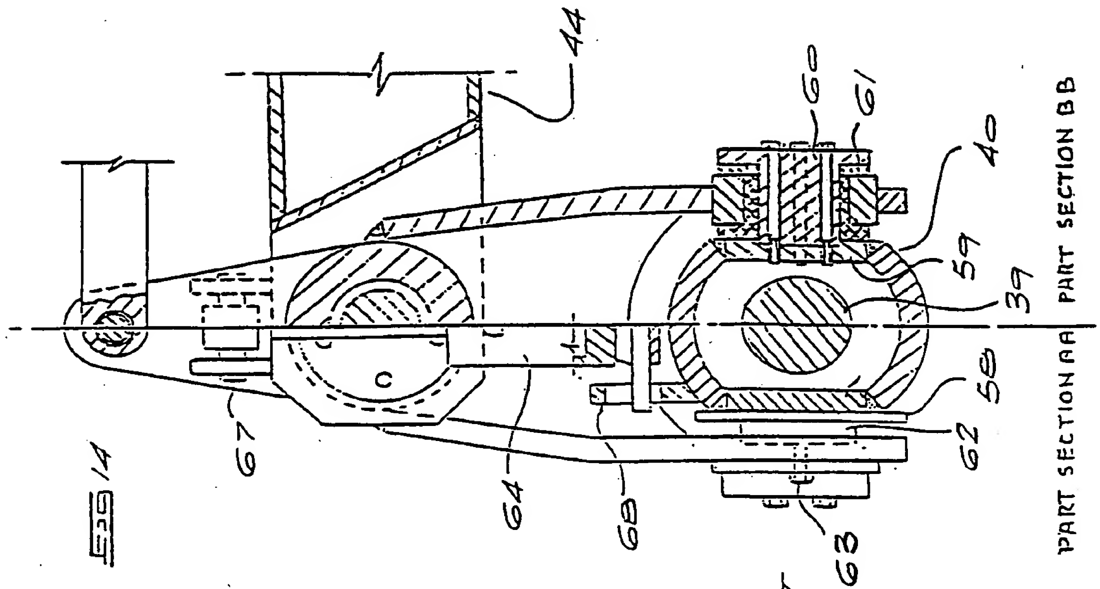


FIG 11

FIG 14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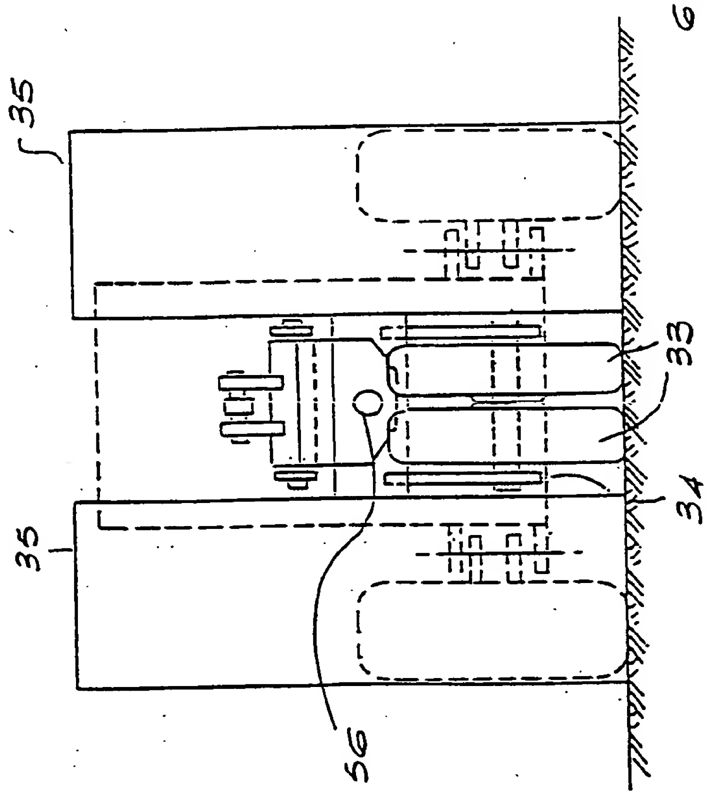
PART SECTION BB

FIG 12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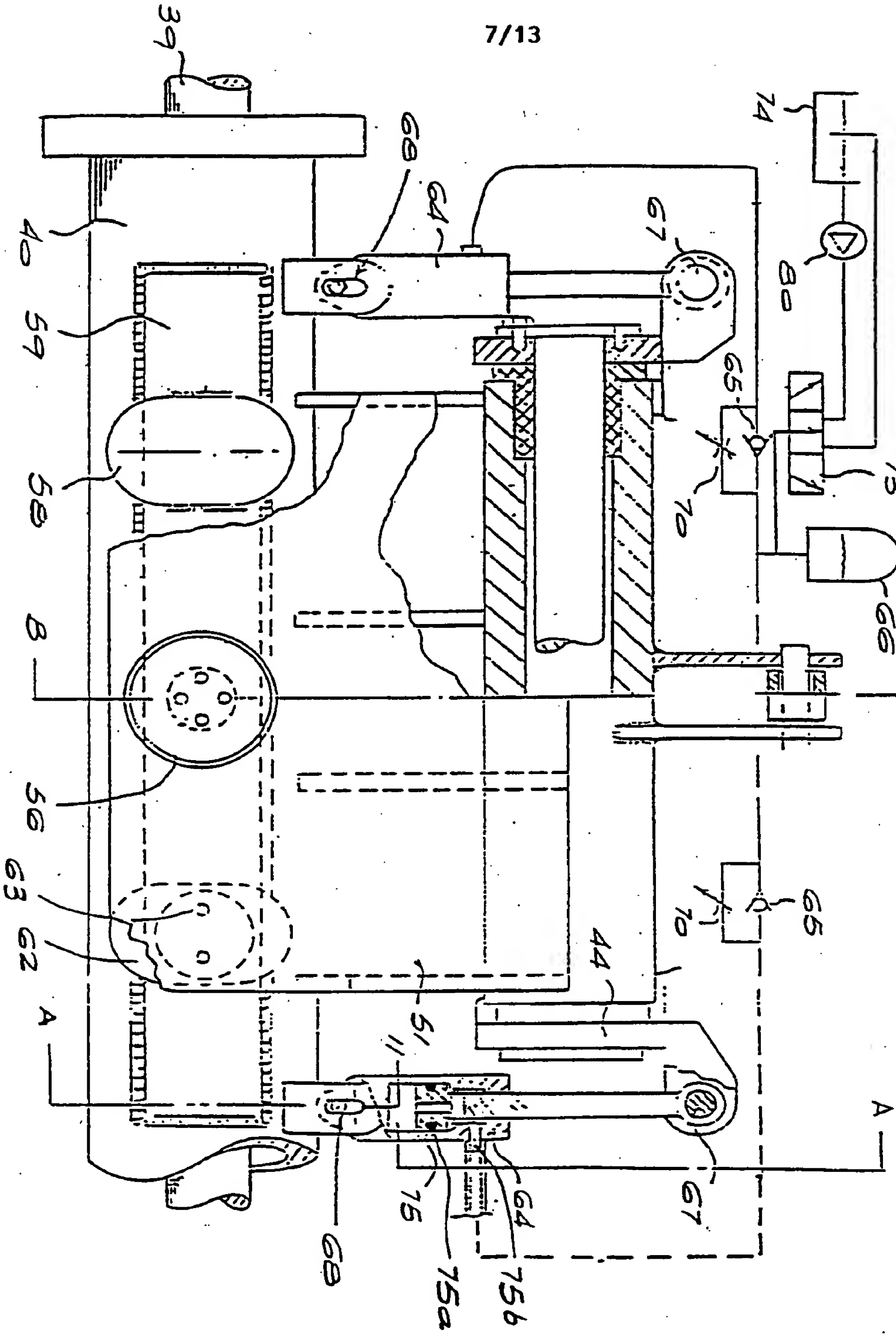
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Fig 13

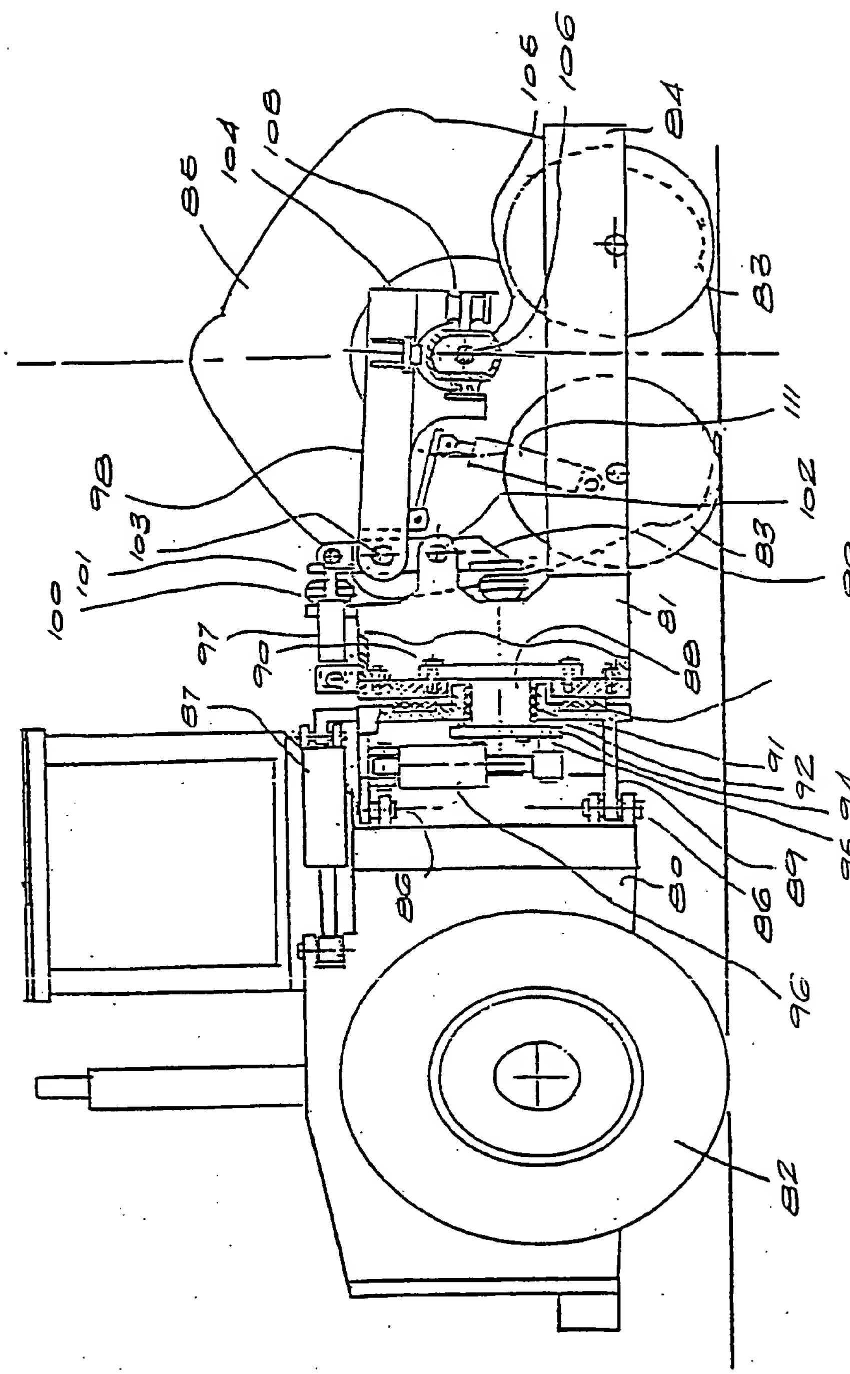
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Fig 15

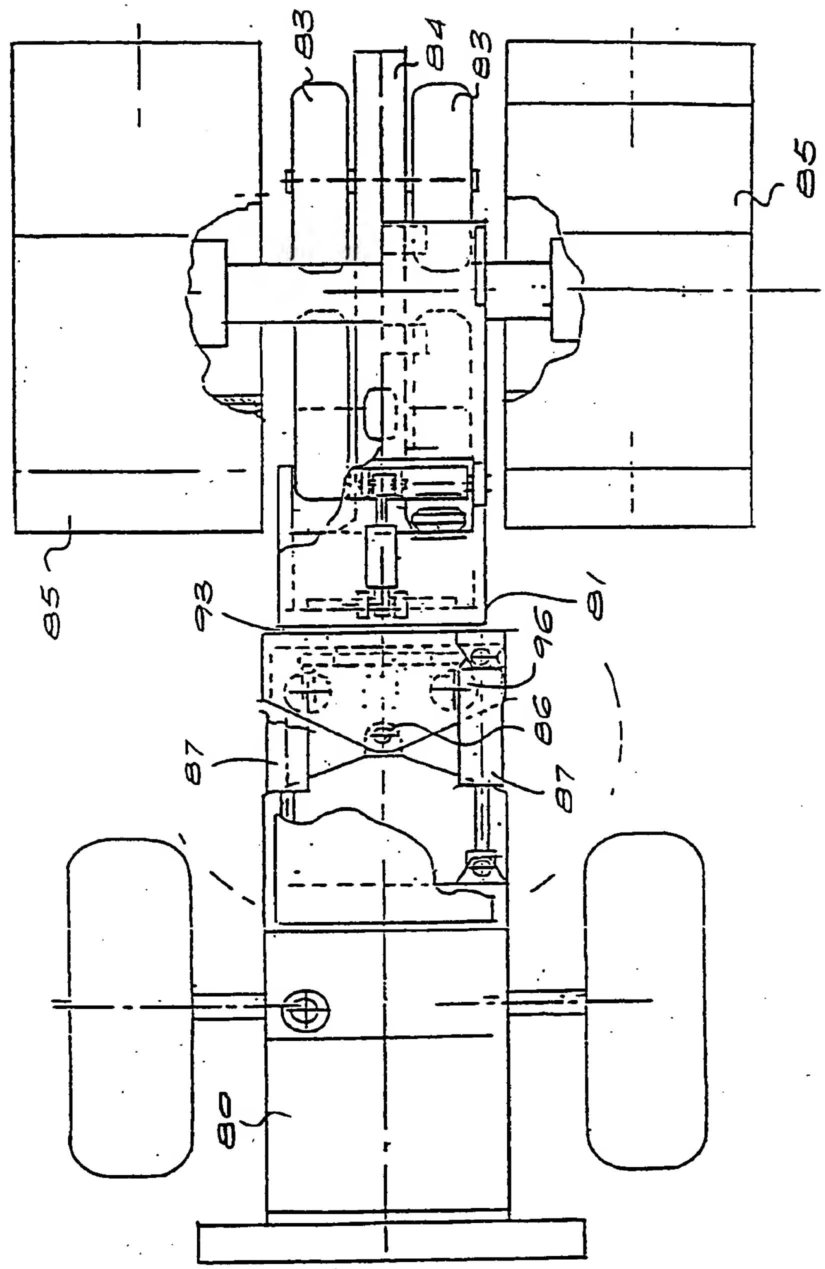

FIG 16

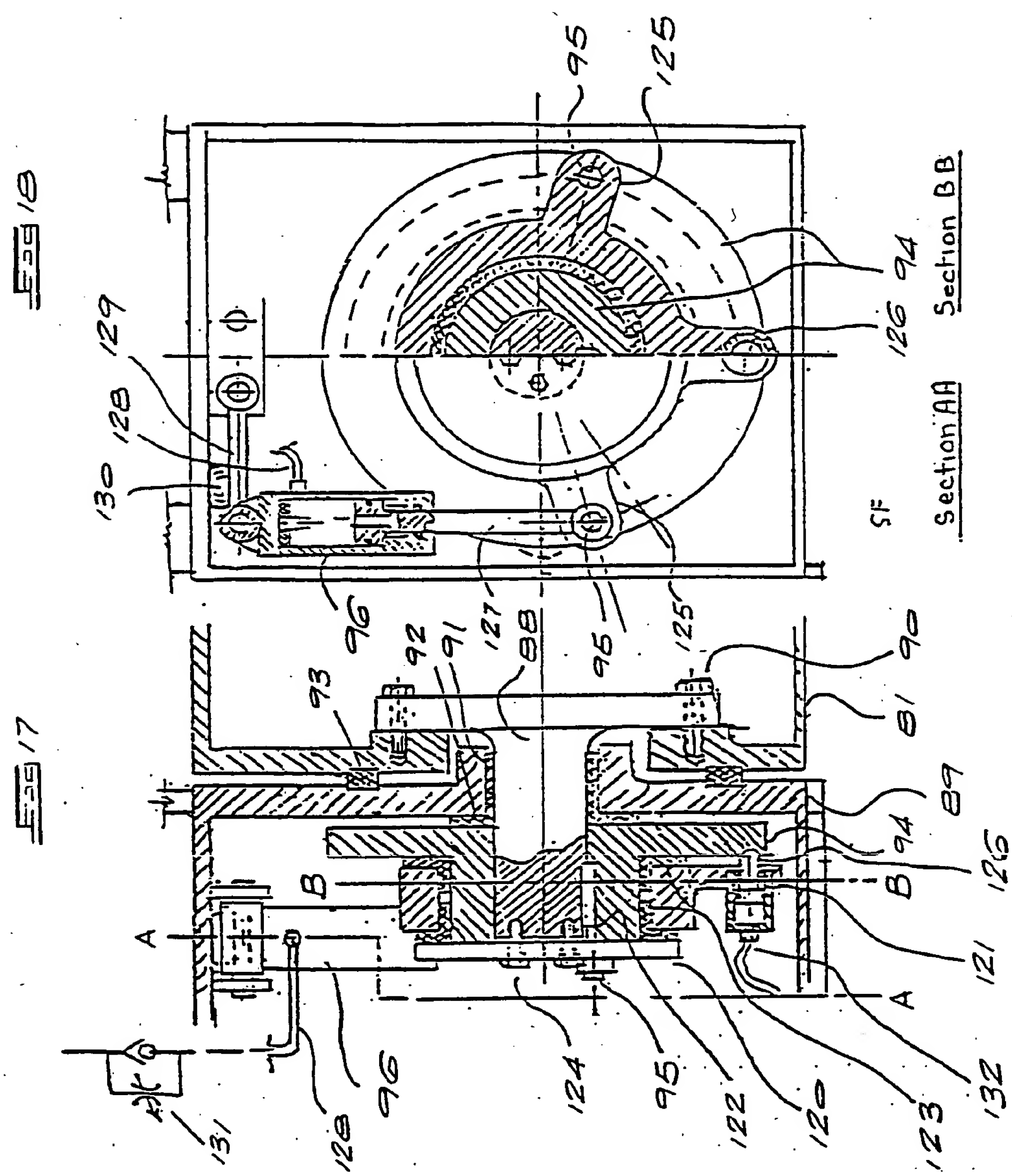
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FIG 19

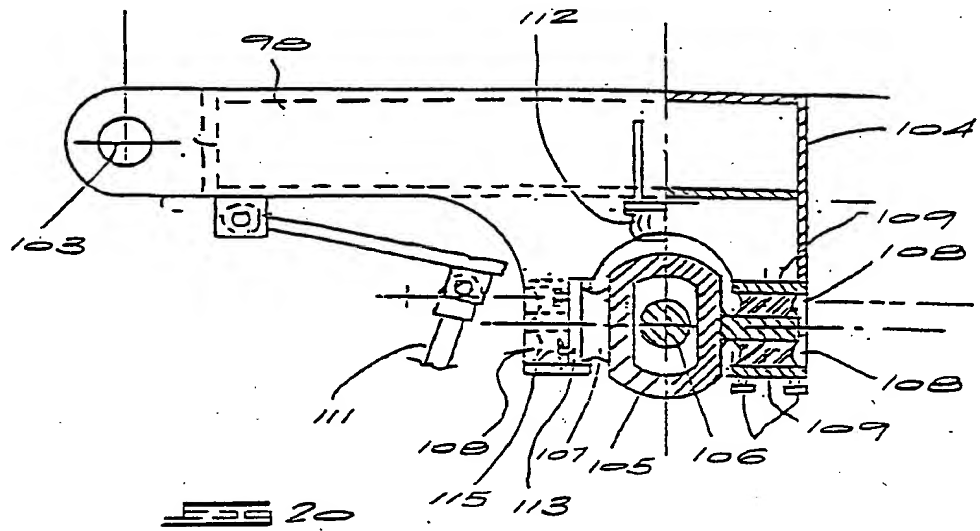


FIG 20

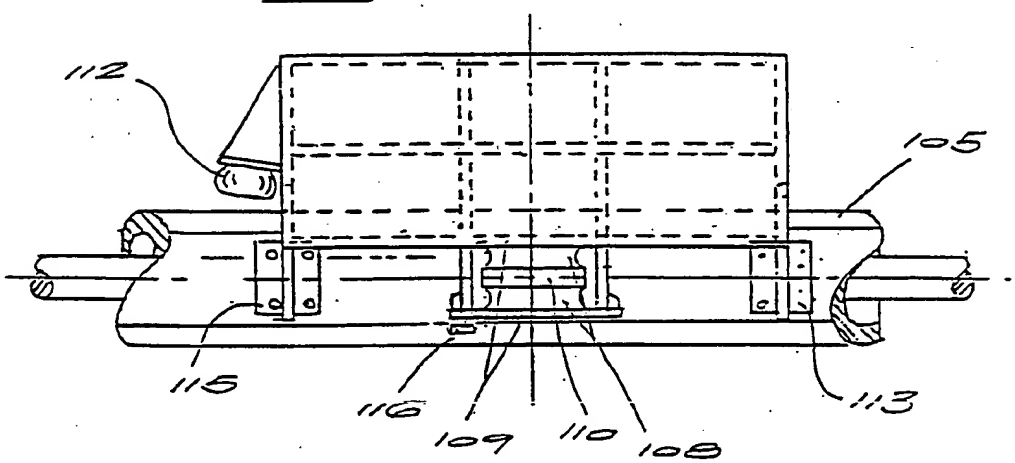


FIG 21

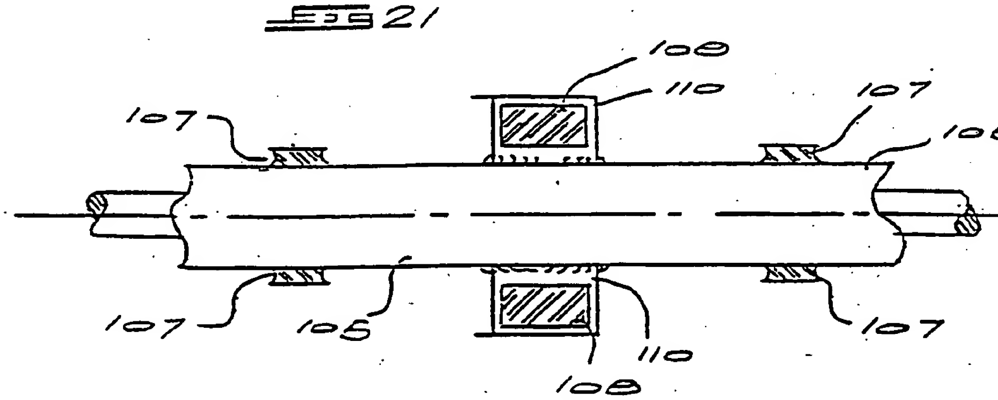


FIG 22

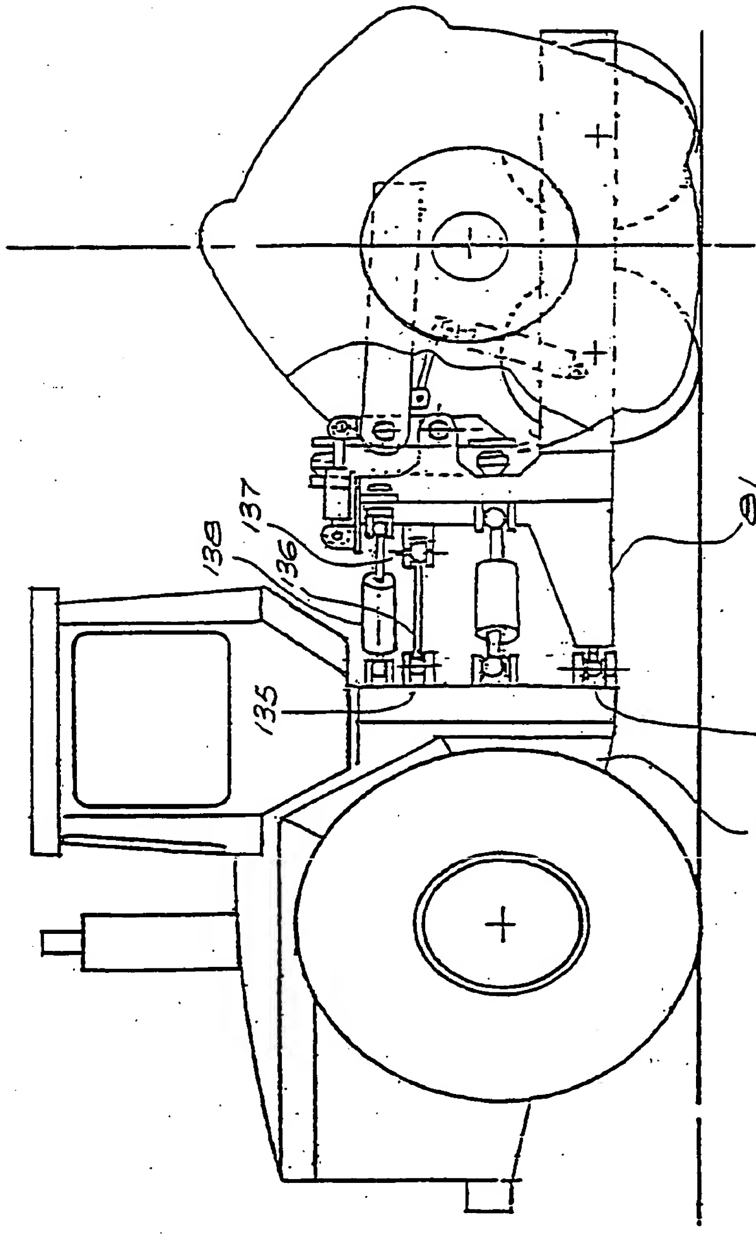
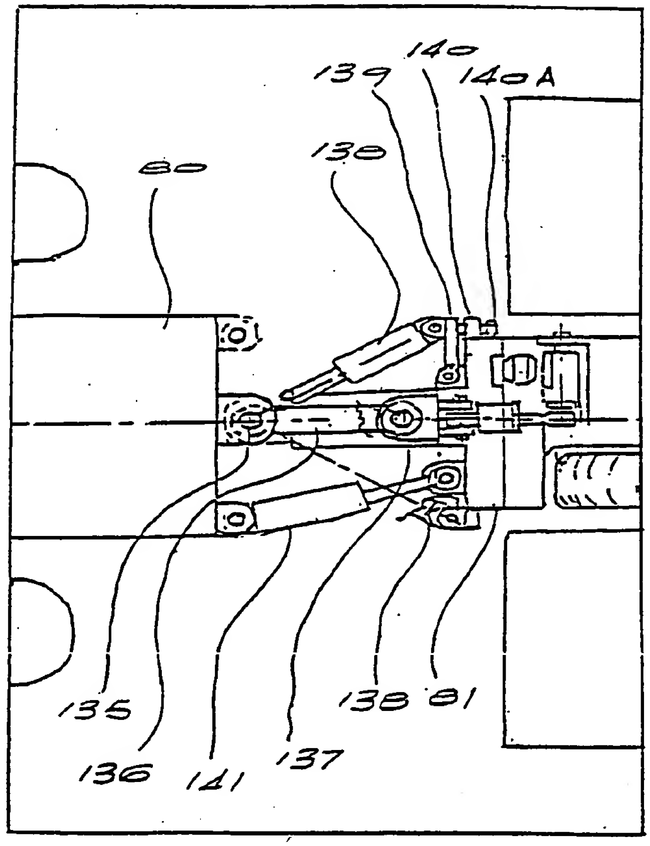


FIG 23



INTERNATIONAL SEARCH REPORT

International Application No
PCT/GB 96/01708

A. CLASSIFICATION OF SUBJECT MATTER
IPC 6 E02D3/026 E01C19/26

According to International Patent Classification (IPC) or to both national classification and IPC

B. FIELDS SEARCHED

Minimum documentation searched (classification system followed by classification symbols)
IPC 6 E02D E01C

Documentation searched other than minimum documentation to the extent that such documents are included in the fields searched

Electronic data base consulted during the international search (name of data base and, where practical, search terms used)

C. DOCUMENTS CONSIDERED TO BE RELEVANT

Category *	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
Y	US,A,3 477 535 (WYATT HAROLD M) 11 November 1969 see the whole document	1-6
Y	EP,A,0 017 511 (BERRANGE AUBREY R) 15 October 1980 cited in the application	1-6
A	see page 1, line 18 - page 4, line 7; figures	11
A	DE,A,28 22 441 (SOUTH AFRICAN INVENTIONS) 7 December 1978 see figures	1,3-5, 7-10
A	DE,A,23 59 375 (SOUTH AFRICAN INVENTIONS) 12 June 1974 see page 1, line 4 - page 3, line 10; figures	1,3-5

☐ Further documents are listed in the continuation of box C.

☒ Patent family members are listed in annex.

* Special categories of cited documents :

"A" document defining the general state of the art which is not considered to be of particular relevance

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"P" document published prior to the international filing date but later than the priority date claimed

"T" later document published after the international filing date or priority date and not in conflict with the application but cited to understand the principle or theory underlying the invention

"X" document of particular relevance; the claimed invention cannot be considered novel or cannot be considered to involve an inventive step when the document is taken alone

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"&" document member of the same patent family

1

Date of the actual completion of the international search	Date of mailing of the international search report
29 October 1996	12.11.96

Name and mailing address of the ISA	Authorized officer
European Patent Office, P.B. 5818 Patentlaan 2 NL - 2280 HV Rijswijk Tel. (+31-70) 340-2040, Tx. 31 651 epo nl, Fax (+31-70) 340-3016	Blommaert, S

INTERNATIONAL SEARCH REPORT

Information on patent family members

International Application No
PCT/GB 96/01708

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		US-A-	4422795	27-12-83
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		JP-A-	54016803	07-02-79
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		SE-A-	7805960	26-11-78
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DE-A-2359375	12-06-74	NONE		